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Theoretical and Experimental Study of an Oil-Free Scroll Vapor Expander

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THEORITICAL AND EXPERIMENTAL STUDY OF AN OIL-FREE SCROLL TYPE VAPOR EXPANDER

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ABSTRACT

The current study investigates experimentally the potential of integrating a scroll expander into a micro cogeneration steam Rankine cycle. This expander technology is not an existing technical option for such applications; consequently an existing oil-free scroll compressor is adapted to run in an expander mode with water as working fluid. The performance of the adapted scroll expander, with dry vapor expansion, is characterized using a test bench developed for this specific application. The tested operating conditions include different operating pressure ratios and different rotation speeds. The results show that the volumetric efficiency increases with the rotation speed, while the isentropic efficiency exhibits an optimum value at a given pressure ratio with a rotation speed of 2000 rpm. A first series of experimentations show a poor volumetric efficiency. The latter was improved by replacing the original tip seal of the expander by a polytetrafluorethylene (PTFE) seal. Experimental results obtained after the modification show 20% increase in the volumetric efficiency without affecting the isentropic efficiency. The increase in volumetric efficiency is mainly due to the lower axial clearance between the seals and the wraps of the expander, while the isentropic efficiency was maintained due to the low leakage flow.

1. INTRODUCTION

In the last years, large efforts have been made to extend the market of micro-CHP (Combined Heat and Power) systems dedicated to produce power and heat in the residential buildings. Small power Rankine cycle producing power and heat presents the advantages of supporting many types of primary energy as fossil fuel or renewable energies as solar energy and biomass. Positive displacement machine can find applications in expander design for micro-CHP systems based on Organic Rankine cycle. Plattel (1993) investigated the potential of screw expanders for the application in small-scale cogeneration, as alternative to turbo expanders. Kane et al. (2003) investigates the use of two superposed Rankine cycles including hermetic scroll expander generator to produce electricity from solar energy and waste heat from diesel engine. Xiaojun et al. (2004) studied the performances of an air scroll expander used for recovering the energy of exhaust high-pressure air from a proton exchange membrane fuel cell. Lemort et al. (2006) studied the integration of a scroll expander into heat recovery Rankine cycle by evaluating experimentally the performances of three different scroll expanders under different operating conditions.

Scroll machines can be operated as expander with high efficiency when operating with air, Yanagisawa et al. (2001). However, a few have shown the performance of these scroll expanders operating with steam, except Lemort et al. (2006) who showed the performance of the scroll expander experimentally for different operating ranges but they did not investigate the effect of the sealing on the performance of the scroll expander. This study presented the possibility to associate a scroll expander to a micro Rankine cycle operating on solar energy or wood boiler working with water. Two different types of seals will be tested to identify the effect of the used sealing type on the performance of the scroll expander.

2. THEORITICAL ANALYSIS

2.1 Scroll expander geometry

The expander selected is originally an oil-free open drive scroll compressor that has been converted to operate in expander mode. The rated power and the nominal flow rate of the compressor are respectively 1.5 kW and 160 L/min at 1920 rpm, 2.2 kW and 240 L/min at 2720 rpm. Its operating efficiency in compression mode has been reported by Yanagisawa *et al.* (1999) and in air expansion mode by Yanagisawa *et al.* (2001). The main dimensions of the expander are: height of wrap: 23.5 mm, thickness of wrap: 4.5 mm, pitch involute wrap: 20.5 mm, involute angles at starting and ending points of wrap: 0.31 and 7.25 π rad. The expander ideal intake and exhaust stroke volumes are 31.5 and 100.1 cm^3/rev, respectively. The built-in volume ratio of the turbine is 3.18 and the built-in expansion ratio is 5.05 for air as a working fluid. The only modification implemented on the original scroll compressor was removing the cooling fan. The high-pressure steam is supplied to the discharge port of the compressor, which leads to the reverse rotation of the machine, namely the turbine operation.

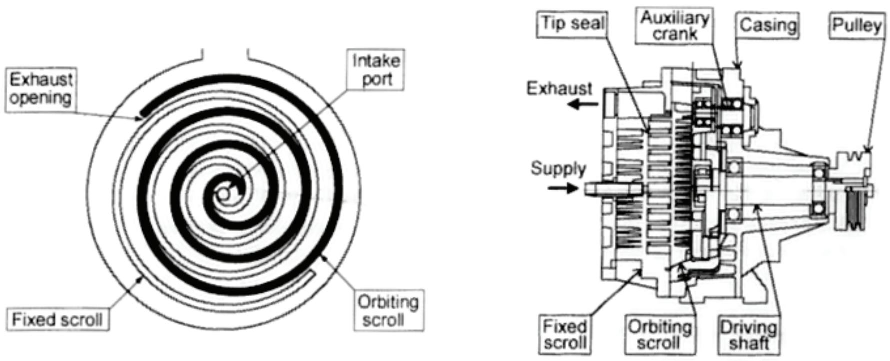


Figure 1: Structure of the experimental scroll expander (Yanagisawa et al. 2001).

2.2 Theoretical performance of the scroll expander

Yanagosawa *et al.* (1999) show that the volumetric and total efficiencies of the compressor are 87% and 56% respectively under the conditions of discharge pressure 700 kPa (gauge) and rotation speed 2720 rpm. On the other hand, Yanagosawa *et al.* (2001) show the performance of the same compressor, but in air expansion mode, to be respectively 76% and 60% for volumetric and total efficiencies, which occur under the conditions of having a pressure supply of 650 kPa and a rotation speed of 2500 rpm. The dominant factor lowering the efficiency was the mechanical losses accompanying the orbiting motion, but the leakage loss through the radial clearance between wraps becomes more significant as the rotation speed decreases.

The theoretical pressure change in the expansion chamber is analyzed taking into account the change of the volume of the chamber. For air, the equation of state of an ideal gas is adopted to calculate the theoretical pressure ratio with a constant adiabatic expansion exponent γ. However, the theoretical expansion ratios of steam in the superheat and in the two-phase regions are calculated with the equation of state resolved in REFPROP 7.0 developed by Lemmon and McLinden (2002). The expansion process is modeled as an isentropic expansion with a constant volume ratio in a closed control volume. The theoretical pressure ratio is calculated by Equation (1).

$$PR = \frac{P_{out}}{P_{in}} = \frac{f\left(\rho_{out}, s_{out}\right)}{P_{in}} = \frac{f\left(\rho_{in}/VR, s_{in}\right)}{P_{in}} \tag{1}$$

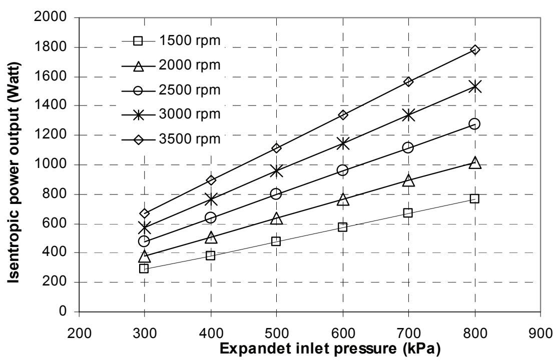
The theoretical mass flow rate depends mainly on the density at the inlet of the scroll expander. The latter depends on the inlet pressure and the temperature of the steam. The theoretical mass flow rate is calculated using Equation (2).

$$\dot{m}_{s,th} = \rho_{s,in} N \dot{V}_{s,th} = \rho_{s,in}\left(T_{s,in}, P_{s,in}\right) N \dot{V}_{s,th} \tag{2}$$

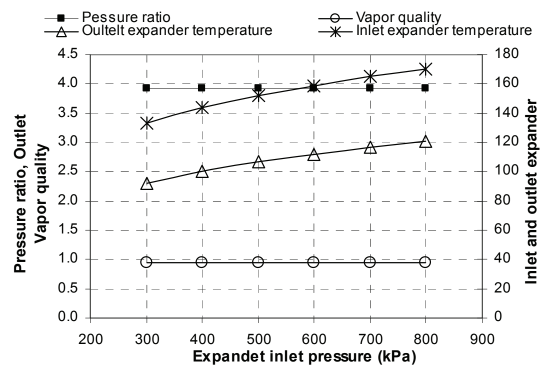
Where N is the rotation speed, $V_{s,th}$ is the chamber volume when the expansion chamber is closed and $\rho_{s,in}$ is the inlet density of the vapor. The isentropic power output can be calculated from an equation based on the first law of thermodynamics for an open control volume with steady state operation and without heat transfer to the surroundings. The isentropic power output of the scroll expander is calculated by Equation (3)

$$W_{is} = \dot{m}_{s,th}\left(h_{turb,in} - h_{turb,out,is}\right) = \dot{m}_{s,th}\left[h\left(P_{in}, T_{in}\right) - h\left[\left(\rho_{in}/VR\right), s_{\text{in}}\right]\right] \tag{3}$$

The theoretical study shows that the ideal pressure ratio of the scroll expander operating with vapor (~3.9) is lower than the ideal pressure ratio with air expansion (~5). Figure 2(b) shows the corresponding inlet and outlet temperatures of the steam for the different boiler pressures. By fixing the condenser pressure at 100 kPa to limit the leakage of steam from the scroll expander to the surroundings at the exit port, the corresponding boiling temperature will be 144°C with a boiling pressure of 400 kPa. Obviously, the highest boiler pressure gives the highest mechanical power. In fact, the vapor density increases with the pressure and thus the MFR expanding via the turbine increases while increasing the boiler pressure. But since the condensing pressure is fixed to 100 kPa, the maximum power output, which could be delivered by the vapor scroll expander, is around 780 W for an inlet pressure of 400 kPa and a rotation speed of 3000 rpm at the ideal pressure ratio of 3.9.



(a)



(b)

Figure 2: (a) Theoretical power output corresponding to different inlet pressures and operating rotation speeds, (b) Evolution of the pressure ratio, outlet vapor quality and turbine outlet temperature.

3. EXPERIMENTAL SETUP

In order to identify the optimum operating parameters of the scroll expander with vapor, a test bench has been realized to test the scroll compressor in expander mode for different rotation speeds, inlet pressures and temperatures.

4.1 Test bench

The steam test bench (figure 3) comprises an electrical heater, a boiler, the expander itself, a condenser and two pumps with variable speed. Moreover, the shaft of the expander is directly coupled to an Eddy current brake. The Eddy current brake imposes the rotation speed of the expander, which can be fixed by an external signal. The electrical heater used in the test bench simulates the heat source. A gear pump circulates a HTF (heat transfer fluid, SYLTHERM 800) through these electrical heaters where it will be heated to a defined temperature selected by the operator; the maximum allowable outlet temperature is 200°C. A power control unit controls the power supplied to the electrical heaters and so the outlet temperature of HTF. A frequency converter controls the speed of the gear pump and so the HTF volumetric flow rate (VFR).

At the outlet of heaters, the HTF flows through the boiler where it exchanges heat with the working fluid (water) of the Rankine cycle. At the outlet of the boiler, the cooled HTF reaches the pump suction to complete the cycle.

A high-pressure diaphragm pump (pump 2) pressurizes and circulates the water via the boiler. A frequency converter controls the speed of the diaphragm pump and so the water VFR. The water is preheated, evaporated and then superheated in the boiler. At the boiler outlet, the superheated steam expands through the turbine and produces mechanical work. The water leaving the turbine is condensed in a water-cooled condenser before reaching the diaphragm pump suction port.

On this test bench, the electrical generator is replaced by an Eddy-current brake (ECB) in order to ensure precise control of the turbine torque and/or speed. This ECB transforms the turbine mechanical work into heat dissipated by a water-cooling circuit.

Temperature and pressure are measured at the inlet and outlet of the scroll expander. The mechanical power output is calculated by measuring simultaneously the rotation speed and the torque developed at the expander shaft by the Eddy current brake. The water VFR is measured by an electromagnetic flow meter. In the experiments, the scroll expander is operating at room temperature under various head pressures and rotation speed conditions.

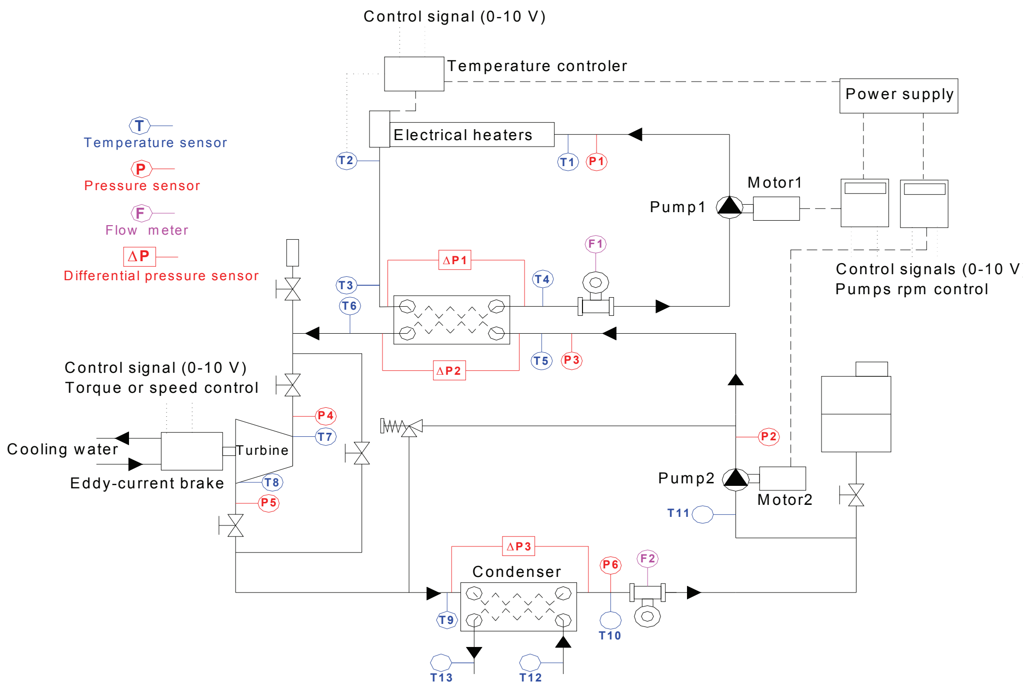


Figure 3: Layout of the test bench.

4.2 Experimental results

The study aims at identifying the operating parameters of a scroll expander associated to a micro-CHP system in order to reach the highest energy performances. First the condenser pressure is fixed to the atmospheric pressure to limit the leakage from the exhaust of the expander to the atmosphere through the tip seals. Another advantage of setting this temperature to 100°C is to ensure the production of co-generated heat at high temperature for heating application. The tests have been performed for boiling pressure of 300 to 500 kPa (water saturation temperature ranging between 133°C and 143°C). The maximum superheat of the steam at the expander inlet is 50 K. The expander rotation speed is limited at 3500 rpm.

The tests are performed to determine the volumetric and isentropic efficiencies of the scroll expander and the mechanical power output. The volumetric efficiency is an important parameter to assess the performance of the expander and is defined as the ratio of the theoretical volume flow rate to the practical volume flow rate as defined by Ziwen (1993).

$$\eta_{vol} = \dot{V}_{s,th} / \dot{V}_{mes} \tag{4}$$

Two primary factors affect the volumetric efficiency. The first parameter is the leakage from the inlet port to the suction chamber, which increases the vapor mass flow rate in the expander. The other factor is throttling effect due to the inlet suction port which results in a charging pressure lower than the nominal suction pressure then a lower vapor mass flow rate enters the expander suction volume. When the effect of leakage exceeds the throttling effect the volumetric efficiency as defined in equation (4) is greater than 100%.

The isentropic efficiency is defined by Equation (5), which represents the real expansion process to the ideal expansion process where there is no resistance loss and no pressure losses during the charging and discharging processes neither losses along the expansion process.

$$\eta_{is} = \left(h_{in,turb} - h_{out,turb}\right) / \left(h_{in,turb} - h_{out,turb,is}\right) \quad (5)$$

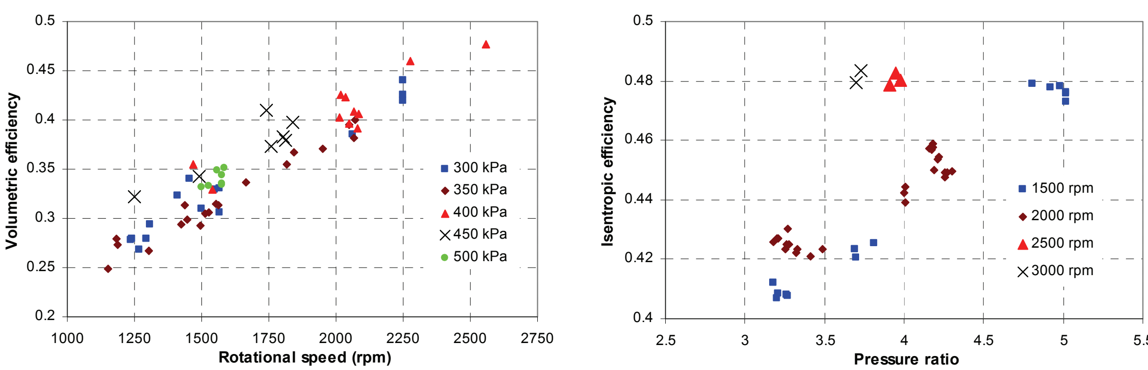
Many parameters affect the isentropic efficiency such as:

- Charging pressure lower than the nominal suction pressure due to the actual pressure losses in the pipes and the expander ports.
- The pressure after expansion could be slightly higher than the condensing pressure: so a part of the steam remains in the discharge pocket and flows backward into the expander.
- Leakage occurs between the wraps and the tips along the expansion process.
- The heat losses from the scroll expander chambers to the surroundings across the body of the expander.

All these factors result in a reduction of the isentropic efficiency.

4.3 Tests and results

Two sets of tests have been performed for the measuring of volumetric and isentropic efficiencies. The first set of tests has been performed with the original gasket of the scroll compressor; the results are presented for the volumetric and the isentropic efficiencies respectively in Figure 4(a) and 4(b). The volumetric efficiency increases gradually with the rotation speed because the leakage flow decreases. The isentropic efficiency increases with the pressure ratio and with the rotation speed as shown in Figure 4(b). For rotation speeds higher than 2500 rpm, the results are limited to some test points because the capacity of the test bench reaches its maximum heating capacity and then it was not possible to increase the pressure ratio or the inlet pressure of the turbine. The maximum isentropic efficiency measured is about 48%, which corresponds to several operating conditions.



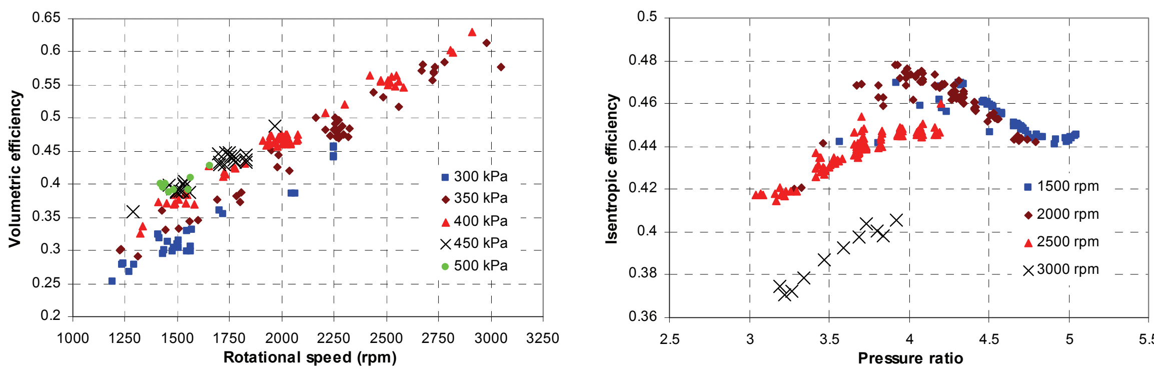
(a) (b)

Figure 4: Evolution of the measured volumetric efficiency (a) and isentropic efficiency (b) for several rotation speeds and for different operating pressure ratios with the original gasket

Since the maximum measured volumetric efficiency (~46%) lies below the predicted efficiency (76%), which is measured with air expansion, the original gasket has been replaced by Polytetrafluoroethylene (PTFE), gasket. The PTFE has been selected since it is adapted for high temperature applications (about 190°C) and has lubricating properties. The PTFE gasket was hand made with a larger width than the original gasket to limit the axial clearance and reduce the leakage flow. The results performed with the new designed gasket are presented in Figures 5(a) and 5(b).

The volumetric efficiency measured with the new designed gasket, shows the same tendency as the previous measurements since it increases while increasing the rotation speed, and exhibits a maximum value of 62% at rotation speed of 2750 rpm and pressure ratio of 4. The results show that the volumetric efficiency has been improved by the replacement of the original gasket by a PTFE gasket to limit the leakage flow rate. The isentropic efficiency exhibits an optimum value for a pressure ratio corresponding to the ideal pressure ratio of the scroll expander and for rotation speed of 2000 rpm.

The improvement of the volumetric efficiency of the scroll expander has allowed extending the measurement of the isentropic efficiency to cover a wider range of operating conditions as shown in Figure 5(b). The tendency of the isentropic efficiency shows that for each operating rotation speed, the optimum isentropic efficiency is close to the ideal expansion ratio, which represents the local optimum. On the other hand, the global optimum isentropic efficiency now occurs at rotation speed of 2000 rpm and expansion ratio of 3.8 near the ideal expansion ratio (~ 4).



(a) (b)

Figure 5: Evolution of the measured volumetric efficiency (a) and isentropic efficiency (b) for several rotation speeds and for different operating pressure ratios with the PTFE gasket.

The results show that the optimum volumetric efficiency (~63%) occurs at 2850 rpm and a pressure ratio of 4, however, the optimum isentropic efficiency (48%) occurs at 2000 rpm and a pressure ratio of 3.8. The isentropic efficiency is less sensitive to the rotation speed compared to the volumetric efficiency. The optimum global efficiency is reached at a pressure ratio of 3.8 and a rotation speed around 2500 rpm with a corresponding volumetric and isentropic efficiency respectively of 55% and 48%.

The mechanical power output measured for the PTFE gasket is presented in Figure 6. At optimum operating conditions, the mechanical power delivered by the expander is about 450 W. However, a higher mechanical power is measured for higher expansion ratio and rotation speed, but these operating points do not correspond to the higher global efficiency. As represented in Figure 6, the maximum power delivered by the expander comes close to 500 W for 3000 rpm and to an expansion ratio of 3.6 but, at these operating conditions, the volumetric and isentropic efficiencies measured are respectively 60% and 38%.

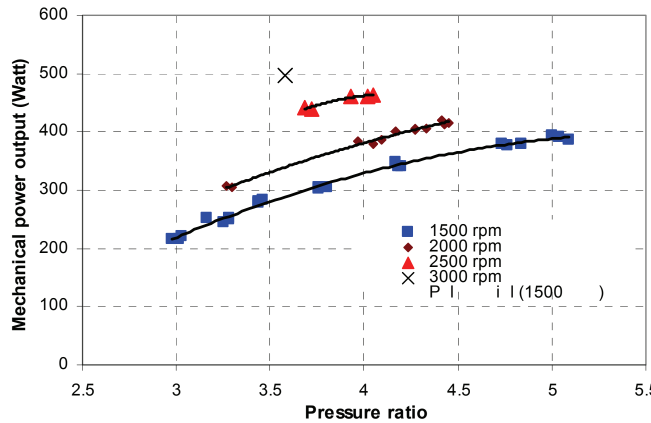


Figure 6: Evolution of the measured mechanical power output with the pressure ratio for different operating rotation speeds with PTFE gasket.

As shown in Figure 6, the turbine mechanical power output tends to increase while increasing the turbine expansion ratio and rotation speed. However, on this test bench, the maximum power for rotation speeds higher than 2500 rpm could not be measured except one point represented for rotation speed of 3000 rpm.

When comparing the performance of the scroll expander with vapor to the same scroll expander operating with air, the performance of the expander is higher for air expansion than for steam expansion. When the expander operates at optimum conditions, the maximum volumetric efficiency achieved is about 62% with vapor and 76% with air. This is mainly due to the steam high operating temperature that results in metal expansion and could lead to larger axial and radial clearances and so larger throttling losses. In addition, for the same operating temperature the steam presents a lower viscosity than air, and therefore larger internal leakages and throttling losses are expected to occur under the same differential pressure.

6. CONCLUSIONS

The performance of an oil-free scroll compressor, which has been converted to operate as oil-free vapor scroll expander, has been evaluated and compared to previous results obtained as expander operating with air expansion. The results of this study are summarized below.

- An oil-free scroll compressor has been successfully converted to operate as oil-free vapor scroll expander at high inlet temperature range between 130°C and 180°C. The main objective of this study is to integrate this scroll expander in micro-cogeneration systems operating on Rankine cycle system.

- The volumetric and isentropic efficiencies of the tested expander were 62% and 48% respectively under the conditions of inlet pressure of 350 kPa, pressure ratio of 3.5 and rotation speed of 2000 rpm.

- The main looses are due to the high leakage flow due to high temperature operation and low viscosity of the steam. Also, the mechanical losses present a major impact on the performance of the expander.

- The main modification applied to the original scroll compressor was the replacement of the original gasket by a PTFE gasket designed to operate at high temperature.

The effect of the viscosity and the temperature of the steam on the performance of the scroll expander have to be evaluated. A physical modeling has to be developed in the future to study the effect of the viscosity on the leakage flow rate and the temperature on the clearance due to the thermal stress deformation.

NOMENCLATURE

PR	Pressure ratio	
P	Pressure	kPa
N	Rotation speed	rpm
VR	Volume ratio	
P	Density	kg/m^3
m	Mass flow rate	kg/s
h	Enthalpy	kJ/kg
s	Entropy	kJ/kg.K
V	Volumetric flow rate	m^3/s
W	Power	kW
T	Temperature	K
η	Efficiency	%

Subscripts

out	outlet
in	inlet
s	suction
th	theoric
turb	turbine
is	isentropic
vol	volumetric
mes	measured

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